Simple adaptive Feedback Controller for the Tower Crane

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Abstract

The objective of crane control is to build an algorithm to move a load from point to point in the shortest time without inducing large swings. We assume that this objective cannot be accomplished in less than a single oscillation cycle of the load. Therefore, the controller is built to move the load such that it completes only one oscillation cycle at the end of the motion. Consequently, the settling time of the system should be equal to the period of oscillation of the load. This criterion enables the calculation of the controller feedback gains for varying load weight and cable length. The controller is built first for the overhead crane and then modified for the tower crane. Two controllers are used, one for the rotational motion of the tower and the other for the translational motion of the trolley. Numerical simulations show that the controller is effective in reducing load oscillations and transferring the load in a reasonable time compared with that of optimal control.

1 Introduction

The operation of the crane can be divided into five steps (Gripping, lifting, moving the load from point to point, lowering, ungripping). A full automation of these processes is possible and there are some researches towards that end[1]. Moving the load from point to point is the most time consuming operation in the process and requires a skillful operator to do this job. Suitable ways to facilitate moving the loads without inducing large swings are the focus of much current research. This work can be classified into the following approaches:

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- Keeping the operator on the loop of the operation and modifying the dynamics of the load to make his job easier. That can be done by:

  1. Adding damping to the swing of the load by feedback from the load swing angle [2].

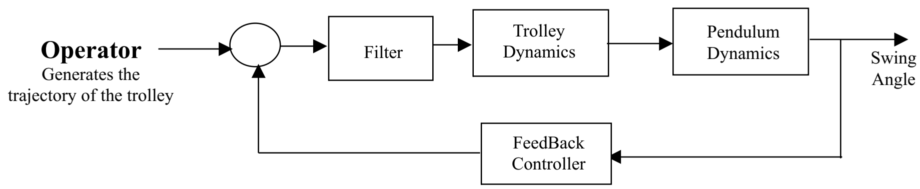


Figure 1: Crane control loop

  2. Avoiding the natural frequency of the load by adding a filter to cut this frequency from the excitation input to the crane, Fig(1) [3].
  3. Adding a mechanical absorber in the structure of the crane [4].

- Moving the load automatically without operator:

  1. Trajectory design methods. Since the swing of the load is affected by the acceleration of the motion trajectory, many researchers concentrated on generating trajectories which minimize the swing especially at the end of the operation with the shortest available time. These trajectories are obtained generally by using optimization techniques [5, 6, 7, 8, 9]. The crucial disadvantage of this method is that it is open loop which makes it sensitive to parameter variations especially if there is a disturbance or change in cable length. One important method of generating the trajectories is the input shaping technique which consists of a sequence of acceleration pulses. These sequences are generated such that there is no residual swing at the end of the transfer operation [10, 11, 12].
  2. Linear feedback control with a step input as a reference trajectory. The gains of the controller are adjusted to make the actual trajectory follow the response of a critically damped system. The root locus technique is usually used for determining the controller gains [13]. A systemic way for determining these gains for varying load weight and cable length is proposed in this paper. This system will be closed loop, hence, it is less affected by parameters variation and disturbance [15, 16, 17].
  3. The third trend is using an optimal trajectory as a reference and increasing the damping in the swing. We end with two control systems, one for tracking the trajectory and the other for increasing the damping in the swing by a proper feedback from the swing angle [18, 19, 20].

Moving the load vertically during transfer (hoisting) is needed only to avoid obstacles. This motion is slow, so, the variation of the rope length can be considered as a disturbance to the system. The effect of the rope length variation is usually studied by simulating the system using the controller designed based on a constant rope length and varying the rope length to make sure the performance does not deteriorate. However, there are few researches which include the hoisting in the design of the controller[5]. The effect of the load weight on the dynamics is usually neglected,h however, for very heavy loads compared to the trolley this effect can not be neglected [22].

Most of the controllers are designed for the gantry crane and there is a few for the tower crane [3, 23]. This research starts by considering a simple crane model which is the gantry crane. The controller parameters are obtained in terms of cable length and load weight for this model then modified to work with the tower crane.

This paper is organized as following, the gantry crane model is derived followed by the control algorithm design. Modeling and the control of the tower crane is then considered. Finally, numerical simulation results are presented and discussed.

2 Gantry Crane Modeling

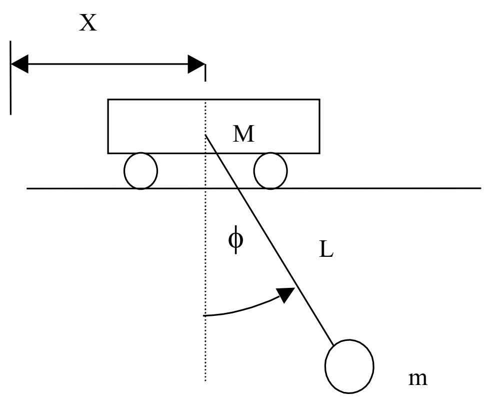


Figure 2: Gantry crane

The Lagrangian approach is used to derive the equations of motion. The load and the trolley position vectors are:

$$\begin{aligned} \vec{r}_L &= \{x + L\sin(\phi)\,,\ L\cos(\phi)\} \\ \vec{r}_T &= \{x, 0\} \end{aligned} \tag{1}$$

The kinetic and potential energies and dissipation function of the whole system are:

$$\begin{aligned}
T &= \frac{1}{2} m\, \dot{\vec{r_L}} \cdot \dot{\vec{r_L}} + \frac{1}{2} M\, \dot{\vec{r_T}} \cdot \dot{\vec{r_T}} \\
V &= -mgL\cos(\phi) \\
D &= \frac{1}{2} b_x \dot{x}^2
\end{aligned} \tag{2}$$

where b_x is the trolley friction coefficient.
Let the generalized forces corresponding to the generalized displacements $\vec{q} = \{x, \theta\}$ to be $\vec{F} = \{F_x, 0\}$. Constructing the Lagrangian $\mathcal{L} = T - V$ and using Lagrange's equation:

$$\frac{d}{dt}\frac{\partial \mathcal{L}}{\partial \dot{q}_j} - \frac{\partial \mathcal{L}}{\partial q_j} + \frac{\partial D}{\partial \dot{q}_j} = F_j, \quad j = 1, ..2 \tag{3}$$

The equations of motion are:

$$\begin{aligned}
&(m + M)\, \ddot{x} + b_x\, \dot{x} + mL\ddot{\phi}\cos(\phi) + m\ddot{L}\sin(\phi) \\
&+ 2m\dot{L}\dot{\phi}\cos(\phi) - mL\dot{\phi}^2\sin(\phi) = F_x
\end{aligned} \tag{4}$$

$$L\ddot{\phi} + g\sin(\phi) + 2\dot{L}\dot{\phi} + \ddot{x}\cos(\theta) = 0 \tag{5}$$

The motor has a small time constant relative to the mechanical system. That enables considering it as a constant gain:

$$F_x = K_\tau V \tag{6}$$

where V is the input voltage to the motor.
During the transfer operation, the swing angle should be kept small. Moreover, changing the cable length is needed only to avoid obstacles in the path of the load. This change can be considered small also. Imposing these two assumptions and dividing equ.(4) by M, the equations of motion reduce to :

$$\ddot{x} + \bar{b}_x \dot{x} - m_t g\phi = \bar{F}_x \tag{7}$$

$$L\ddot{\phi} + g\phi + \ddot{x} = 0 \tag{8}$$

where:

$$\bar{b}_x = \frac{b_x}{M}, m_t = \frac{m}{M}, \bar{F}_x = \frac{F}{M} \tag{9}$$

For simplicity, the bar will be omitted further.

3 Control Algorithm

A simple state feedback controller can be used for controlling the position of the trolley and reducing the swing. The first proposed controller has the form:

$$V = K\left(x_{ref} - K_{px}x - K_{dx}\dot{x} + K_{pa}\phi\right) \tag{10}$$

The question that arises now is how to adjust these gains to get the best performance for a wide range of cable length and load. The main objective is to make the swing as small as possible. The minimum number load oscillation that can be achieved is one cycle. It is noted that when the trolley response is critically damped, the load completes one oscillation cycle. That indicates that the settling time for the trolley should be equal to the time period of the load. That gives a good criterion to choose the location of the closed loop poles and hence the feedback gains [21]. The design procedure will be as following.
To make the trolley response critically damped, choose its poles to be repeated and equal to $-a$. The load performance should have oscillatory behavior, then make its poles to be $-\zeta\,\omega_n \pm \omega_n\sqrt{1-\zeta^2}j$. The final closed loop Characteristic equation will be:

$$(s+a)^2\left(s^2 + 2\zeta\,\omega_n s + \omega_n^2\right) = 0 \tag{11}$$

From simulation, the best damping ratio which can be chosen is $\zeta = \dfrac{1}{\sqrt{2}}$. Using the settling time criterion mentioned above:

$$\frac{4}{a} = \frac{2\pi}{\sqrt{g/L}} \quad \rightarrow \quad a = \frac{4\sqrt{g/L}}{2\pi} \tag{12}$$

Comparing the required closed loop characteristic equation (11) with that of the system, four nonlinear equations will be obtained . The steady state error is given by: $A\left(1 - \dfrac{1}{K_{px}}\right)$, where A is value of the step input to the system. To have zero steady state error, let $K_x = 1$. The rest of the system parameters $(b_x,\, K_\tau)$ are known and fixed. We end up with four equations in four unknowns $(K, K_{dx}, K_{pa}, \omega_n)$. These variables can be calculated symbolically as functions of $(m_t,\, L)$. At this stage, there is no control over K which is the dominant factor in determining the maximum acceleration of the trolley especially at the beginning of motion.
The acceleration increases as the error increases. So, for long travel distance, the motor acceleration required at the beginning will be very high which is not realistic. Also, this acceleration increases as L decreases because it will be required to move the load to its target in short time which requires high speed and consequently high acceleration at the beginning. To overcome the previous problem, K could be predetermined for not exceeding the maximum acceleration of the sytem. Now, we end with four equations in four unknowns $(K_x,\, K_a,\, K_t, \omega_n)$ which can be calculated as before as functions of $(m_t,\, L,\, K)$. The problem here is the steady state error which will not be zero. To make it zero, another gain should

be added. This leads to the second proposed controller which is a full state feedback:

$$V = K\left(x_{ref} - K_{px}x - K_{dx}\dot{x} + K_{pa}\phi + K_{da}\dot{\phi}\right) \tag{13}$$

Applying the same principle by defining the characteristic equation as (11). If choose K$_x$= 1 to get zero steady state error and choose K for not exceeding the maximum available control action, we can get (K_{dx}, K_{pa}, K_{da}, ω_n) as functions of the remaining parameters (m_t, L).

3.1 Stability

The stability of the system is governed by ω_n. For asymptotic stability, it should be positive which is guaranteed when the gains are calculated.

4 Tower Crane Modeling and Control

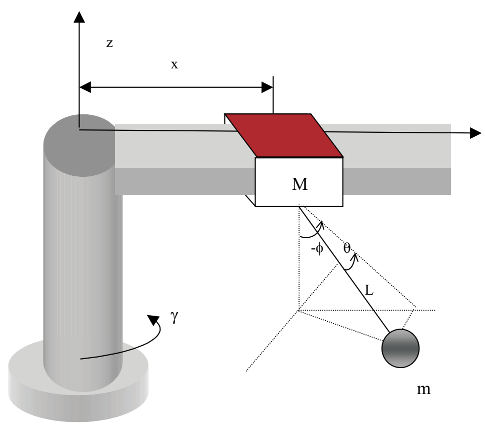


Figure 3: Tower crane

As in the gantry modeling, the Lagrangian approach will be used to derive the equations of motion of the tower crane. The load and trolley position vectors can be written as:

$$\begin{aligned} \vec{r}_L &= \{x - L\cos(\theta)\sin(\phi)\,,\ L\sin(\theta)\,,\ -L\cos(\theta)\cos(\phi)\} \\ \vec{r}_T &= \{x,\ ,0,0\} \end{aligned} \tag{14}$$

The velocities of the trolley and the load are:

$$\dot{\vec{r}} = \frac{d\vec{r}}{dt} + \vec{\omega}\times\vec{r} \tag{15}$$

where $\vec{\omega} = \{0, 0, \dot{\gamma}\}$ is the angular velocity of the tower.
The kinetic and potential energies and the dissipation function are:

$$\begin{aligned} T &= \frac{1}{2}m\,\vec{\dot{r}}_L \cdot \vec{\dot{r}}_L + \frac{1}{2}M\,\vec{\dot{r}}_T \cdot \vec{\dot{r}}_T + \frac{1}{2}J_o\dot{\gamma}^2 \\ V &= -mgL\cos(\theta)\cos(\phi) \\ D &= \frac{1}{2}b_x\dot{x}^2 + \frac{1}{2}b_\gamma\dot{\gamma}^2 \end{aligned} \tag{16}$$

where J_o is the moment of inertia of the tower and the jib about z axis. b_x and b_γ are the friction coefficients for the trolley and the tower respectively.
The generalized forces corresponding to the generalized displacement $\vec{q} = \{x, \phi, \gamma, \theta\}$ are:

$$\vec{F} = \{F_x, 0, T, 0\} \tag{17}$$

Constructing the Lagrangian $\mathcal{L} = T - V$ and using Lagrange's equation:

$$\frac{d}{dt}\frac{\partial \mathcal{L}}{\partial \dot{q}_j} - \frac{\partial \mathcal{L}}{\partial q_j} + \frac{\partial D}{\partial \dot{q}_j} = F_j, \quad j = 1, ..4 \tag{18}$$

The equations of motion are nonlinear and complicated. These equations will be used for simulation. However for analysis, we need to simplify them. For small swing angles and neglecting the variation in the cable length, the equations of motion reduce to:

$$M\ddot{x} + b_x\dot{x} + mg\phi + Mx\dot{\gamma}^2 = F_x \tag{19}$$

$$L\ddot{\phi} + 2L\dot{\gamma}\dot{\theta} - L\dot{\gamma}^2\phi + L\ddot{\gamma}\theta - \ddot{x} - x\dot{\gamma}^2 + g\phi = 0 \tag{20}$$

$$(J_o + Mx^2)\ddot{\gamma} + b_\gamma\dot{\gamma} - mgx\theta + 2mxL\dot{\gamma}^2\theta + mL(\dot{x}\dot{\gamma})\phi + (m + 2M)x\dot{x}\dot{\gamma} = T \tag{21}$$

$$L\ddot{\theta} - 2L\dot{\gamma}\dot{\phi} + g\theta - L\dot{\gamma}^2\theta - L\ddot{\gamma}\phi + x\ddot{\gamma} + \dot{x}\dot{\gamma} = 0 \tag{22}$$

The above equations are still complicated for designing the controller. For further simplification, it is assumed that the rate of change of x and γ have the same order of magnitude as the swing rate and the change in the length. The above equations after dividing by M and J_o and neglecting the low order nonlinear terms will be reduced to:

$$\ddot{x} + \bar{b}_x\dot{x} + m_t g\phi = \bar{F}_x \tag{23}$$

$$L\ddot{\phi} + g\phi - \ddot{x} = 0 \tag{24}$$

$$(1 + M_r x^2)\ddot{\gamma} + \bar{b}_\gamma\dot{\gamma} - m_r g x\theta = \bar{T} \tag{25}$$

$$L\ddot{\theta} + g\theta + x\ddot{\gamma} = 0 \tag{26}$$

where:

$$\bar{b}_x = \frac{b_x}{M}, \bar{b}_\gamma = \frac{b_x}{J_o}, m_t = \frac{m}{M}, M_r = \frac{M}{J_o}, \bar{F}_x = \frac{F}{M}, \bar{T} = \frac{T}{Jo} \tag{27}$$

For simplicity, the bar will be omitted further. The two motors can be modelled as constant gains:

$$\bar{F} = K_{\tau x} V_t \tag{28}$$

$$\bar{T} = K_{\tau\gamma} V_r \tag{29}$$

Equations (23-26) can be divided into two groups. Translational motion equations (23,24) and rotational motion (25,26). It is noticed that the translational motion equations can be derived from the rotational one by putting $x = -1$ and $M_r = 0$.

4.1 Tower Crane Control

The translational equations are similar to the gantry crane equations. The same controller can be used for controlling this motion. The rotational motion equations are coupled with the translation ones by the trolley position. This coupling can be relaxed by assuming x to be constant at any instant. That enables using the same control techniques used for the gantry. Moreover, the feedback gains will be varied with the trolley position.

5 Simulation Results

The full nonlinear equations are used for the simulation with the following numerical values for the crane system:

$$b_x = b_\gamma = 2.65, \quad K_\tau = K_{\tau x} = K_{\tau\gamma} = 1.34, \quad M_t = 0.5$$

The time is scaled by the oscillation period of the load: $\bar{t} = \dfrac{t}{2\pi/\sqrt{g/L}} = \dfrac{t}{T}$. This scaling makes the responses similar for all values of L.

5.1 Gantry Crane

Variations of the feedback gains with cable length for $m_t = 0$ and the load mass for $L = 0$ using the reduced state feedback controller are shown in Fig.(4). We can notice that K is inversely proportional to the cable length because the load natural frequency increases with the length which decreases the system settling time and consequently increases K. So, It is better to raise the load as much as possible to transfer it in a short time. However, the speed of the trolley should not exceed the motor maximum speed. The swing angle gain is linearly proportional L. If the swing distance $L\phi$ is used in the feedback instead of ϕ, K_{pa} would be a constant with varying L. It is shown also that the swing angle gain reduces linearly with m_t while the other gains do not change. Examining equations (4,5), we can notice that $m_t g\phi$ plays as an extra force to to the trolley equation which affects the trolley acceleration and consequently the swing angle. The change of K_{pa} with m_t can be considered a compensation for this extra force. The effect of changing the load mass on the performance of the system are shown in Fig.(5). From inspection of the time histories, it is seen that the system response deteriorates if the gains are not adapted for the change in the load mass. An overshoot and hence an increase in the number of load oscillations occur for using gains calculated for $m_t = 0$ to control the system with $m_t = 5$ because k_{pa} will be higher than the required. We can notice the same performance when the cable length is changed without adapting the gains as shown in Fig.(6). The deterioration increases with increasing cable length and load weight. The performance is sensitive to the length variation more than the mass. The responses due to the implementation of the second controller for different values of K are shown in Fig.(7). The control action and the swing decreases with decreasing K while the response becomes slightly slower.

5.2 Tower Crane

In these simulations, $m_t = 1$ and hence $m_r = M_r * m_t = 0.5$. The response for $L = 1$ is shown in Fig.(9). This response shows the similarity between tower and gantry cranes. In Fig.(10), the cable length is changed to $L = 5$ while the gains are calculated for $L = 1$. It is obvious that the response deteriorates if the gains are not adapted with the variation of L. The response using full state feedback with $K = 0.4$ and $L = 1$ is shown in Fig.(11). It is also noticed that the swing decreases with decreasing K. At the same time, the response becomes slower because of the decrease in the control authority. It was found by simulation that to improve the response the gains should only be changed with the trolley position if it exceeds one. For trolley positions less than one the gains are kept constant at their values at $x = 1$. The response to disturbance to the load position is shown in Fig.(12) which indicates that the disturbance is effectively damped. However, when reduced state feedback is used, the oscillations associated θ will not die. This shows the effect of the nonlinearity in the system which should be included in the design of the controller to improve the response.

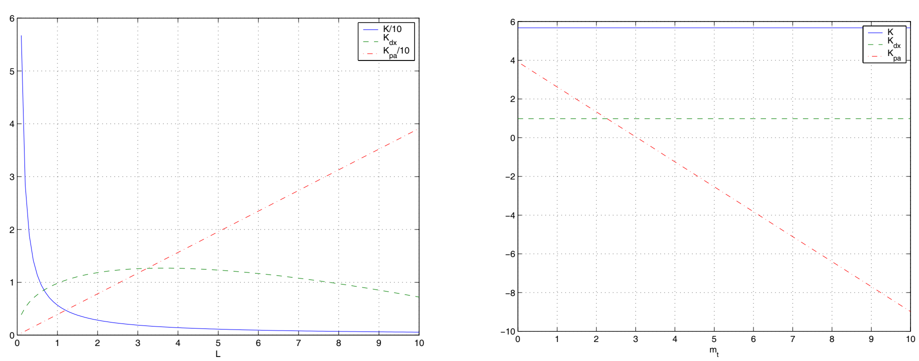
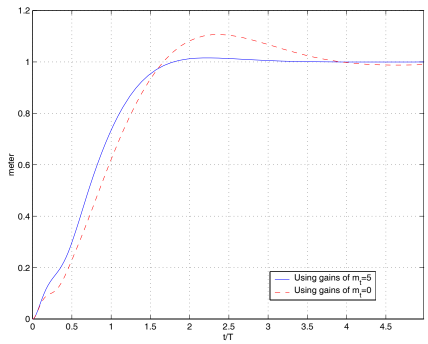
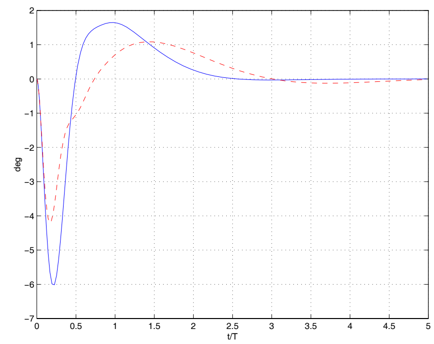


Figure 4: Variations of feedback gains with m_t and L

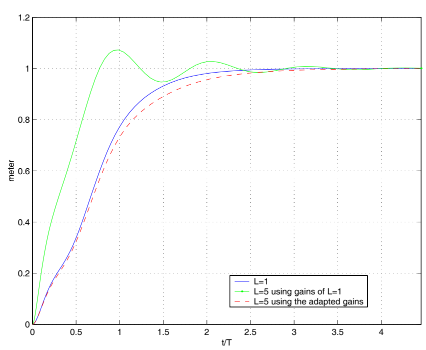


(a) Trolley position x

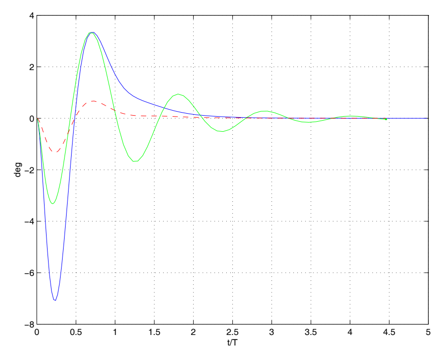


(b) Load swing ϕ

Figure 5: Effect of the changing load weight $m_t = 5$, $L = 1$

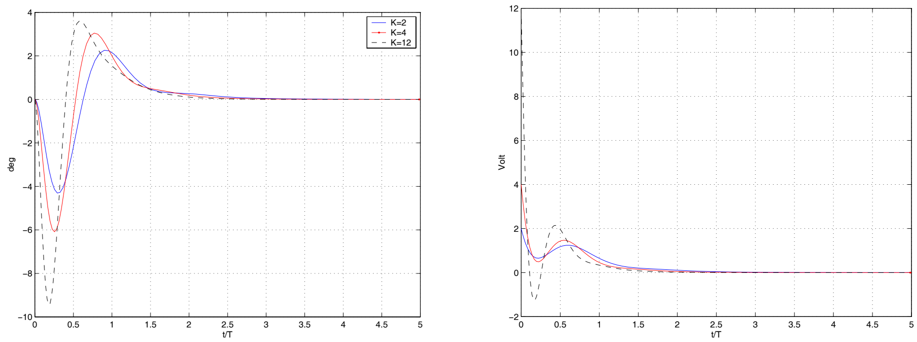


(a) Trolley position x



(b) Load swing ϕ

Figure 6: Effect of changing cable length from $L = 1$ to $L = 5$



(a) Load swing ϕ

(b) Control action

Figure 7: Effect of changing k using full state feedback, $L = 1$, $m_t = 0$

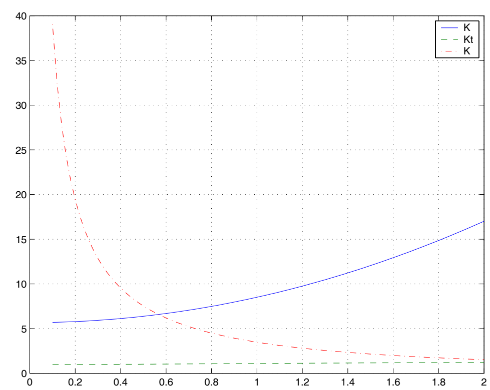
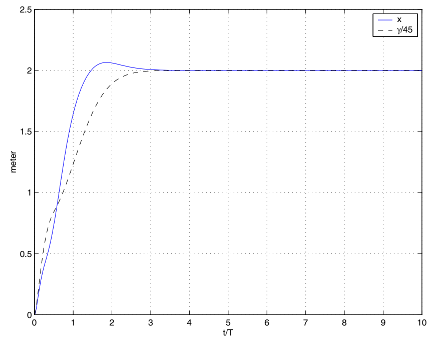
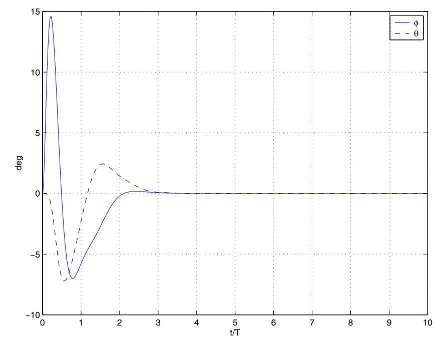


Figure 8: Variations of the feedback gains with x for $M_r = m_r = 0.5$

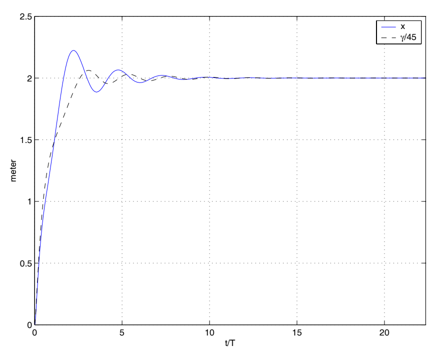


(a) Trolley and tower positions

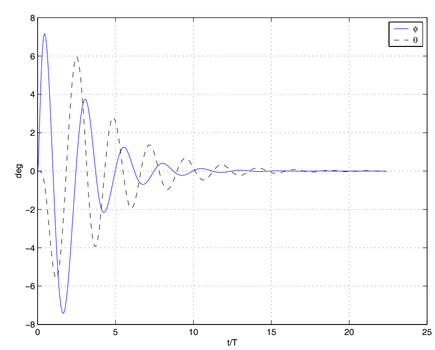


(b) Load swing ϕ and θ

Figure 9: Tower crane: $L = 1$, $m_t = .5$

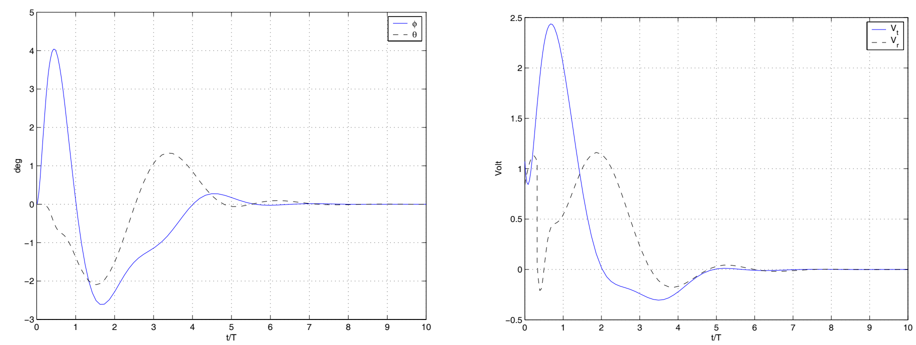


(a) Trolley and tower positions



(b) Load swing

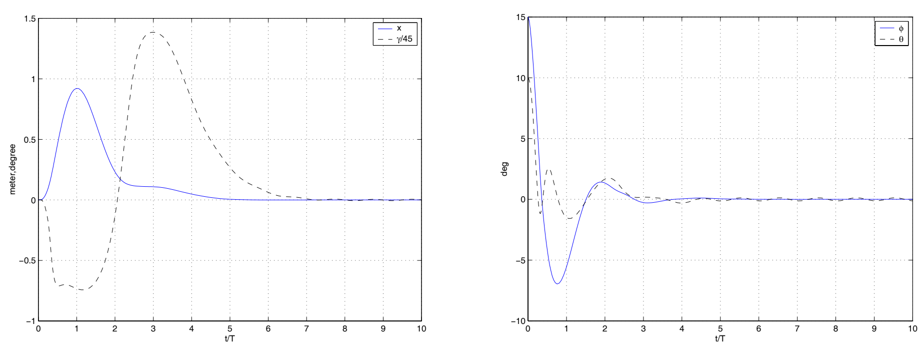
Figure 10: Tower crane: $L = 5$ using the gains calculated for $L = 1$



(a) Load swing

(b) Control action

Figure 11: Tower crane: $L = 1$ and $K = 0.4$



(a) Trolley and tower positions

(b) Load swing

Figure 12: Tower crane: initial disturbance $\phi = 10^o, \theta = 5^o$ with $K = 0.4, L = 1, m_r = .5$

6 Summary

The state feedback can be used to control the position and reduce the swing in the tower crane with wide range of cable length and load weight. This controller is closed loop which makes it suitable for disturbance rejection and insensitive to the changes in the system. The step input command is very severe input because it generates high acceleration at the beginning of the motion. This input should be replaced by an trajectory obtained from solving the time optimal control of the tower crane. Moreover, the system performance could be improved if the nonlinearities are considered in the design of the controller which will be the next stage in this research.

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